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APPLICATION TO RIGID MEMORY MECHANISMS OF A VARIABLE INTERNAL DYNAMIC DAMPING MODEL

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ABSTRACT

The paper presents a dynamic model that works with variable internal damping, applicable directly to rigid memory mechanisms. If the problem of elasticity is generally solved, the problem of system damping is not clear and well-established. It is usually considered a constant "c" value for the internal damping of the system and sometimes the same value c and for the damping of the elastic spring supporting the valve. However, the approximation is much forced, as the elastic spring damping is variable, and for the conventional cylindrical spring with constant elasticity parameter (k) with linear displacement with force, the damping is small and can be considered zero. It should be specified that damping does not necessarily mean stopping (or opposition) movement, but damping means energy consumption to brake the motion (rubber elastic elements have considerable damping, as are hydraulic dampers). Metal helical springs generally have a low (negligible) damping. The braking effect of these springs increases with the elastic constant (the k-stiffness of the spring) and the force of the spring (P0 or F0) of the spring (in other words with the arc static arrow, x0=P0/k). Energy is constantly changing but does not dissipate (for this reason, the yield of these springs is generally higher). The paper presents a dynamic model with a degree of freedom, considering internal damping of the system (c), damping for which it is considered a special function..

More precisely, the cushioning coefficient of the system (c) is defined as a variable parameter depending on the reduced mass of the mechanism (m* or J reduced) and the time, ie, c depends on the derivative of m reduced in time. The equation of the differential movement of the mechanism is written as the movement of the valve as a dynamic response.

Keywords: Distribution mechanism; Rigid memory mechanisms; Variable internal damping; Dynamic model; Angular speed variation; Dynamic coefficient

1. INTRODUCTION

The development and diversification of road vehicles and general vehicles, especially of cars, together with thermal engines, especially internal combustion engines (being more compact, robust, more independent, more reliable, stronger, more dynamic etc.)., has also forced the development of devices, mechanisms, and component assemblies at an alert pace. The most studied are power and transmission trains.

The four-stroke internal combustion engine (four-stroke, Otto or Diesel) comprises in most cases (with the exception of rotary motors) and one or more camshafts, valves, valves, and so on.

The classical distribution mechanisms are robust, reliable, dynamic, fast-response, and although they functioned with very low mechanical efficiency, taking much of the engine power and effectively causing additional pollution and increased fuel consumption, they could not be abandoned until the present. Another problem was the low speed from which these mechanisms begin to produce vibrations and very high noises.

Regarding the situation realistically, the mechanisms of cam casting and sticking are those that could have produced more industrial, economic, social revolutions in the development of mankind. They have contributed substantially to the development of internal combustion engines and their spreading to the detriment of external combustion (Steam or Stirling) combustion engines.

The problem of very low yields, high emissions and very high power and fuel consumption has been greatly improved and regulated over the past 20-30 years by developing and introducing modern distribution mechanisms that, besides higher

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yields immediately deliver a high fuel economy also performs optimal noise-free, vibration-free, no-smoky operation, as the maximum possible engine speed has increased from 6000 to 30000 [rpm].

The paper tries to provide additional support to the development of distribution mechanisms so that their performance and the engines they will be able to further enhance.

Particular performance is the further increase in the mechanical efficiency of distribution systems, up to unprecedented quotas so far, which will bring a major fuel economy.

The current oil and energy reserves of mankind are limited. Until the implementation of new energy sources (to take real control over fossil fuels), a real alternative source of energy and fuel is even "the reduction in fuel consumption of a motor vehicle", whether we burn oil, gas and petroleum derivatives, whether we will implement biofuels first, and later hydrogen (extracted from water).

The drop in fuel consumption for a given vehicle type over a hundred kilometers traveled has been consistently since 1980 and has continued to continue in the future.

Even if hybrids and electric motor cars are to be multiplied, let us not forget that they have to be charged with electricity, which is generally obtained by burning fossil fuels, especially oil and gas, in a current planetary proportion of about 60%. Can burn oil in large heat plants to warm up, have domestic hot water and electricity to consume, and some of that energy is extra and we add it to electric cars (electric vehicles), but the global energy problem is not resolved, the crisis even deepens. This was the case when was electrified the railroad for trains, when it were generalized trams, trolleybuses and subways, consuming more electric power produced mainly from oil; oil consumption has grown a lot, its price has had a huge leap, and now one looks at how the reserves disappear quickly.

Generally, generalizing electric cars (though it is not really ready for this), one will give a new blow to oil and gas reserves.

Fortunately, biofuels, biomass and nuclear power have developed very much lately (currently based on the nuclear fission reaction). These together with the hydroelectric power plants have managed to produce about 40% of the total energy

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consumed globally. Only about 2-3% of global energy resources are produced by various other alternative methods (despite the efforts made so far).

This should not disarm us, and abandon the implementation of solar, wind, etc.

However, as a first necessity to further reduce the share of global energy from oil and gas, the first vigorous measures that will need to be pursued will be to increase biomass and biofuels production along with the widening of the number of nuclear power plants (despite some undesirable events, which only show that nuclear fission power plants must be built with a high degree of safety, and in no way eliminated from now on, and they are still the one that has been so far "a bad evil ").

Alternative sources will take them on an unprecedented scale, but it expects the energy they provide to be more consistent in global percentages so that can rely on them in a real way (otherwise, one risks that all these alternative energies remain a sort of "fairy tale").

Hydrogen fuel energy "when it starts when it stops" so there is no real time now to save energy through them, so they can no longer be priority, but the trucks and buses could even be implemented now that the storage problems have been partially solved. The bigger problem with hydrogen is no longer the safe storage, but the high amount of energy needed to extract it, and especially for its bottling.

The huge amount of electricity consumed for bottling hydrogen will have to be obtained entirely through alternative energy sources, otherwise hydrogen programs will not be profitable for humanity at least for the time being. The authors thinking the immediate use of hydrogen extracted from the water with alternative energies would be more appropriate for seagoing vessels.

Maybe just to say that due to his energy crisis (and not just energy, from 1970 until today), the production of cars has increased at an alert pace (but naturally) instead of falling, and they have and were marketed and used. The world's energy crisis (in the 1970s) began to rise from around 200 million vehicles worldwide, to about 350 million in 1980 (when the world's energy and global fuel crisis was declared), about 500 million vehicles worldwide, and in 1997 the number of world-registered vehicles exceeded 600 million.

In 2010, more than 800 million vehicles circulate across the planet (ANTONESCU, 2000; ANTONESCU; PETRESCU, 1985; ANTONESCU; PETRESCU,

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1989; ANTONESCU et al., 1985a; ANTONESCU et al.,1985b; ANTONESCU et al., 1986; ANTONESCU et al., 1987; ANTONESCU et al., 1988; ANTONESCU et al., 1994; ANTONESCU et al., 1997; ANTONESCU et al., 2000a; ANTONESCU et al.2000b ANTONESCU et al., 2001; AVERSA et al., 2017a; AVERSA et al., 2017b; AVERSA et al., 2017c; AVERSA et al., 2017d; AVERSA et al., 2017e; MIRSAYAR et al., 2017; PETRESCU et al., 2017a; PETRESCU et al., 2017b; PETRESCU et al., 2017c; PETRESCU et al., 2017d; PETRESCU et al., 2017e; PETRESCU et al., 2017f; PETRESCU et al., 2017g; PETRESCU et al., 2017h; PETRESCU et al., 2017i, PETRESCU et al.,2015; PETRESCU; PETRESCU, 2016; PETRESCU; PETRESCU, 2014; PETRESCU; PETRESCU, 2013a; PETRESCU; PETRESCU, 2013b; PETRESCU; PETRESCU, 2013c; PETRESCU; PETRESCU, 2013d; PETRESCU; PETRESCU, 2011; PETRESCU; PETRESCU, 2005a; PETRESCU; PETRESCU, 2005b; PETRESCU, 2015a; PETRESCU, 2015b, PETRESCU, 2012a; PETRESCU, 2012b; HAIN, 1971; GIORDANA et al., 1979; ANGELES; LOPEZ-CAJUN, 1988; TARAZA et al., 2001; WIEDERRICH; ROTH, 1974; FAWCETT; FAWCETT, 1974; JONES; REEVE, 1974; TESAR; MATTHEW, 1974; SAVA, 1970; KOSTER, 1974).

2. THE STATE OF THE ART

The Peugeot Citroën Group in 2006 built a 4-valve hybrid engine with 4 cylinders the first cam opens the normal valve and the second with the phase shift. Almost all current models have stabilized at four valves per cylinder to achieve a variable distribution. In 1971, K. Hain proposes a method of optimizing the cam mechanism to obtain an optimal (maximum) transmission angle and a minimum acceleration at the output. In 1979, F. Giordano investigates the influence of measurement errors in the kinematic analysis of the camel.

In 1985, P. Antonescu presented an analytical method for the synthesis of the cam mechanism and the flat barbed wire, and the rocker mechanism. In 1988, J. Angeles and C. Lopez-Cajun presented the optimal synthesis of the cam mechanism and oscillating plate stick.

In 2001 Dinu Taraza analyzes the influence of the cam profile, the variation of the angular speed of the distribution shaft and the power, load, consumption and emission parameters of the internal combustion engine. In 2005, Petrescu and

Petrescu, present a method of synthesis of the rotating camshaft profile with rotary or rotatable tappet, flat or roller, in order to obtain high yields at the exit.

In the paper (WIEDERRICH; ROTH, 1974), there is presented a basic, single-degree, dual-spring model with double internal damping for simulating the motion of the cam and punch mechanism. In the paper (FAWCETT; FAWCETT, 1974) is presented the basic dynamic model of a cam mechanism, stick and valve, with two degrees of freedom, without internal damping.

A dynamic model with both damping in the system, external (valve spring) and internal one is the one presented in the paper (JONES; REEVE, 1974). A dynamic model with a degree of freedom, generalized, is presented in the paper (TESAR; MATTHEW, 1974), in which there is also presented a two-degree model with double damping.

In the paper (SAVA, 1970) is proposed a dynamic model with 4 degrees of freedom, obtained as follows: the model has two moving masses these by vertical vibration each impose a degree of freedom one mass is thought to vibrate and transverse, generating yet another degree of freedom and the last degree of freedom is generated by the torsion of the camshaft.

Also in the paper (SAVA, 1970) is presented a simplified dynamic model, amortized. In (SAVA, 1970) there is also showed a dynamic model, which takes into account the torsional vibrations of the camshaft. In the paper (KOSTER, 1974) a four-degree dynamic model with a single oscillating motion mass is presented, representing one of four degrees of freedom.

The other three freedoms result from a torsional deformation of the camshaft, a vertical bending (z), camshaft and a bending strain of the same shaft, horizontally (y), all three deformations, in a plane perpendicular to the axis of rotation. The sum of the momentary efficiency and the momentary losing coefficient is 1. The work is especially interesting in how it manages to transform the four degrees of freedom into one, ultimately using a single equation of motion along the main axis.

The dynamic model presented can be used wholly or only partially, so that on another classical or new dynamic model, the idea of using deformations on different axes with their cumulative effect on a single axis is inserted. In works (ANTONESCU et al., 1987; PETRESCU; PETRESCU, 2005a) there is presented a dynamic model

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with a degree of freedom, considering the internal damping of the system (c), the damping for which is considered a special function. More precisely, the damping coefficient of the system (c) is defined as a variable parameter depending on the reduced mass of the mechanism (m* or $J_{reduced}$) and time, ie, c, depends on the time derivative of $m_{reduced}$. The equation of differential movement of the mechanism is written as the movement of the valve as a dynamic response.

3. DYNAMIC MODELS WITH VARIABLE INTERNAL DAMPING

Starting from the kinematic scheme of the classical distribution mechanism (see Figure 1), the dynamic, mono-dynamic (single degree), translatable, variable damping model (see Figure 2) is constructed, the motion equation of which is:

$$M \cdot \ddot{x} = K \cdot (y - x) - k \cdot x - c \cdot \dot{x} - F_0 \quad (1)$$

Equation (1) is nothing else than the equation of Newton, in which the sum of forces on an element in a certain direction (x) is equal to zero.

The notations in formula (1) are as follows:

M- mass of the reduced valve mechanism;

K- reduced elastic constants of the kinematic chain (rigidity of the kinematic chain);

k- elastic spring valve constant;

c - the damping coefficient of the entire kinematic chain (internal damping of the system);

F, F_t - the elastic spring force of the valve spring;

x - actual valve displacement;

(the cam profile) reduced to the axis of the valve.

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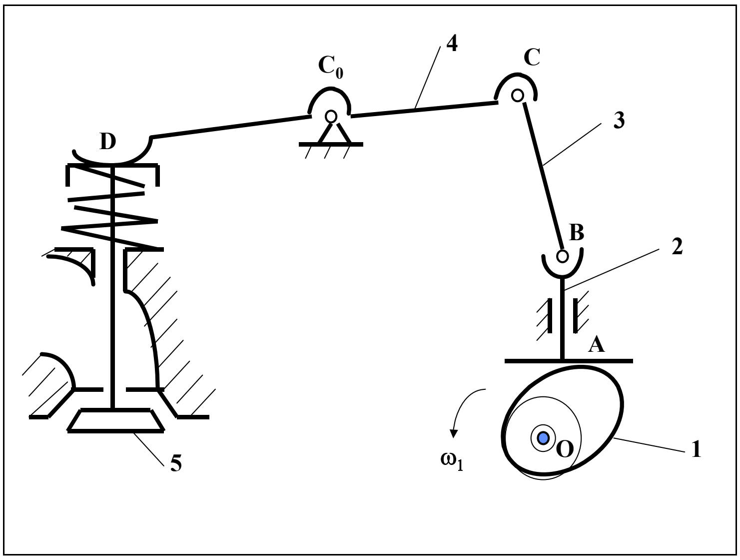

Figure 1: The kinematic scheme of the classic distribution mechanism

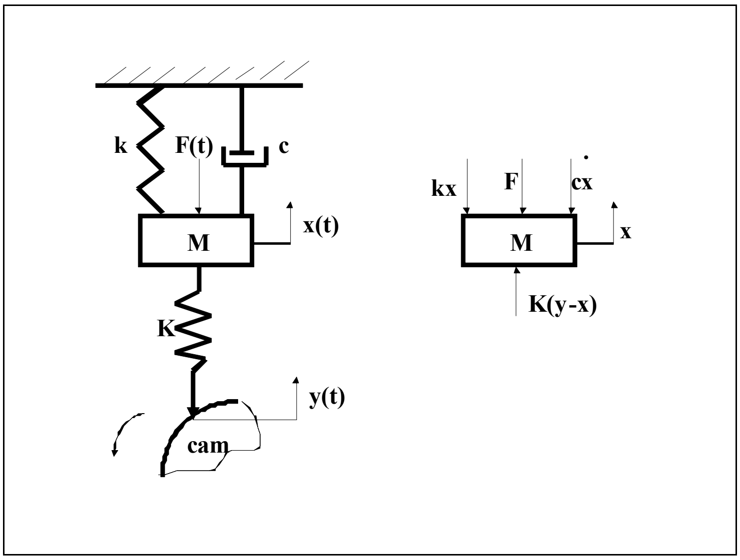

Figure 2: Mono - dynamic model, with internal depreciation of the variable system

The Newton equation (1) is ordered as follows:

$$M \cdot \ddot{x} + c \cdot \dot{x} = K \cdot (y - x) - (F_0 + k.x) \tag{2}$$

At the same time the differential equation of the mechanism is also written as Lagrange, (3), (Lagrange equation):

$$M \cdot \ddot{x} + \frac{1}{2}\frac{dM}{dt} \cdot \dot{x} = F_m - F_r \tag{3}$$

Equation (3), which is nothing other than the Lagrange differential equation, allows for the low strength of the valve (4) to be obtained by the polynomial coefficients with those of the Newtonian polynomial (2), the reduced drive force at the valve (5),

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as well as the expression of c, ie the expression of the internal damping coefficient, of the system (6).

$$F_r = F_0 + k \cdot x = k \cdot x_0 + k \cdot x = k \cdot (x_0 + x) \quad (4)$$

$$F_m = K \cdot (y - x) = K \cdot (s - x) \quad (5)$$

$$c = \frac{1}{2} \cdot \frac{dM}{dt} \quad (6)$$

Thus a new formula (6) is obtained, in which the internal damping coefficient (of a dynamic system) is equal to half the derivative with the time of the reduced mass of the dynamic system.

The Newton motion equation (1, or 2), by replacing it with c takes the form (7):

$$M \cdot \ddot{x} + \frac{1}{2}\frac{dM}{dt} \cdot \dot{x} + (K + k) \cdot x = K \cdot y - F_0 \quad (7)$$

In the case of the classical distribution mechanism (in Figure 1), the reduced mass, M, is calculated by the formula (8):

$$M = m_5 + (m_2 + m_3) \cdot \left(\frac{\dot{y}_2}{\dot{x}}\right)^2 + J_1 \cdot \left(\frac{\omega_1}{\dot{x}}\right)^2 + J_4 \cdot \left(\frac{\omega_4}{\dot{x}}\right)^2 \quad (8)$$

formula in which or used the following notations:

m_2 = stick weight;

m_3 = the mass of the pushing rod;

m_5 = mass of the valve;

J_1 = moment of mechanical inertia of the cam;

J_4 = moment of mechanical inertia of the culbutor;

$\dot{y}_2$ = velocity of stroke imposed by cam law;

$\dot{x}$ = valve speed.

If i = i_{25}, the valve-to-valve ratio (made by the crank lever), the theoretical velocity of the valve (imposed by the motion law given by the cam profile) is calculated by the formula (9):

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$$y \equiv \dot{y}_5 = \frac{\dot{y}_2}{i} \tag{9}$$

where:

$$i = \frac{CC_0}{C_0 D} \tag{10}$$

is the ratio of the crank arms.

The following relationships are written (11-16):

$$\dot{x} = \omega_1 \cdot x' \tag{11}$$

$$\ddot{x} = \omega_1^2 \cdot x'' \tag{12}$$

$$\dot{y}_2 = \omega_1 . y_2' = \omega_1 . i . y' \tag{13}$$

$$\frac{\omega_1}{\dot{x}} = \frac{\omega_1}{\omega_1 . x'} = \frac{1}{x'} \tag{14}$$

$$\omega_4 = \frac{\dot{y}_2}{CC_0} = \frac{\omega_1 . y_2'}{CC_0} = \frac{\omega_1 . y' . i}{CC_0} = \frac{\omega_1 . y'}{CC_0} \frac{CC_0}{C_0 D} = \frac{\omega_1 . y'}{C_0 D} \tag{15}$$

$$\frac{\omega_4}{\dot{x}} = \frac{\omega_1 . y'}{C_0 D . \omega_1 . x'} = \frac{1}{C_0 D} \frac{y'}{x'} \tag{16}$$

where y 'is the reduced velocity imposed by the camshaft (by the law of camshaft movement), reduced to the valve axis.

With the previous relationships (10), (13), (14), (16), the relationship (8) becomes (17-19):

$$M = m_5 + (m_2 + m_3) \cdot (\frac{i . y'}{x'})^2 + J_1 \cdot (\frac{1}{x'})^2 + J_4 \cdot (\frac{1}{C_0 D} \frac{y'}{x'})^2 \tag{17}$$

or:

$$M = m_5 + [i^2 \cdot (m_2 + m_3) + \frac{J_4}{(C_0 D)^2}] \cdot (\frac{y'}{x'})^2 + J_1 \cdot (\frac{1}{x'})^2 \tag{18}$$

or:

$$M = m_5 + m^* \cdot (\frac{y'}{x'})^2 + J_1 \cdot (\frac{1}{x'})^2 \tag{19}$$

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We make the derivative dM/d and result the following relationships:

$$\frac{d[(\frac{y'}{x'})^2]}{d\varphi} = \frac{2 \cdot y'}{x'} \cdot \frac{(y''\cdot x' - x''\cdot y')}{x'^2} = \frac{2 \cdot y'}{x'^2} \cdot (y'' - x'' \cdot \frac{y'}{x'}) = 2 \cdot (\frac{y'}{x'})^2 \cdot (\frac{y''}{y'} - \frac{x''}{x'}) \quad (20)$$

$$\frac{d[(\frac{1}{x'})^2]}{d\varphi} = \frac{2}{x'} \cdot \frac{-x''}{x'^2} = -2 \cdot \frac{x''}{x'^3} \quad (21)$$

$$\frac{dM}{d\varphi} = 2 \cdot m^* \cdot (\frac{y'}{x'})^2 \cdot (\frac{y''}{y'} - \frac{x''}{x'}) - 2 \cdot J_1 \cdot \frac{x''}{x'^3} \quad (22)$$

Write the relationship (6) as:

$$c = \frac{\omega}{2} \cdot \frac{dM}{d\varphi} \quad (23)$$

which with (22) becomes:

$$c = \omega \cdot \{[i^2 \cdot (m_2 + m_3) + \frac{J_4}{(C_0 D)^2}] \cdot (\frac{y'}{x'})^2 \cdot (\frac{y''}{y'} - \frac{x''}{x'}) - J_1 \cdot \frac{x''}{x'^3}\} \quad (24)$$

or

$$c = \omega \cdot [m^* \cdot (\frac{y'}{x'})^2 \cdot (\frac{y''}{y'} - \frac{x''}{x'}) - J_1 \cdot \frac{x''}{x'^3}] \quad (25)$$

Where was noted:

$$m^* = i^2 \cdot (m_2 + m_3) + \frac{J_4}{(C_0 D)^2} \quad (26)$$

4. DETERMINATION OF MOTION EQUATIONS

With relations (19), (12), (25) and (11), equation (2) is written first in the form (27), which develops in forms (28), (29) and (30):

$$M \cdot \omega^2 \cdot x'' + c \cdot \omega \cdot x' + (K + k) \cdot x = K \cdot y - F_0 \quad (27)$$

$$\begin{cases} \omega^2 \cdot x'' \cdot m_5 + \omega^2 \cdot m^* \cdot (\frac{y'}{x'})^2 \cdot x'' + J_1 \cdot (\frac{1}{x'})^2 \cdot x'' \cdot \omega^2 + \omega^2 \cdot x' \cdot m^* \cdot (\frac{y'}{x'})^2 \cdot (\frac{y''}{y'} - \frac{x''}{x'}) - \\ x' \cdot \omega^2 \cdot J_1 \cdot \frac{x''}{x'^3} + (K + k) \cdot x = K \cdot y - F_0 \end{cases} \quad (28)$$

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meaning:

$$\omega^2 \cdot m_5 \cdot x''+\omega^2 \cdot m^* \cdot x'' \cdot (\frac{y'}{x'})^2 - \omega^2 \cdot m^* \cdot (\frac{y'}{x'})^2 \cdot x''+\omega^2 \cdot m^* . y'' \cdot \frac{y'}{x'} + (K+k) \cdot x = K \cdot y - F_0 \quad (29)$$

And final form:

$$\omega^2 \cdot m_5 \cdot x''+(K+k) \cdot x + \omega^2 \cdot m^* \cdot y'' \cdot \frac{y'}{x'} = K \cdot y - F_0 \quad (30)$$

which can also be written in another form:

$$\omega^2 \cdot (m_5 \cdot x''+m^* \cdot y'' \cdot \frac{y'}{x'}) + (K+k) \cdot x = K \cdot y - F_0 \quad (31)$$

Equation (31) can be approximated to form (32) if we consider the theoretical input velocity y imposed by the camshaft profile (reduced to the valve axis) approximately equal to the velocity of the valve, x.

$$\omega^2 \cdot (m_5 \cdot x''+m^* \cdot y'') + (K+k) \cdot x = K \cdot y - F_0 \quad (32)$$

If the laws of entry with s, s' (low speed), s' '(low acceleration), equation (32) takes the form (33) and the more complete equation (31) takes the complex form (34):

$$\omega^2 \cdot (m_5 \cdot x''+m^* \cdot s'') + (K+k) \cdot x = K \cdot s - F_0 \quad (33)$$

$$\omega^2 \cdot (m_5 \cdot x''+m^* \cdot s'' \cdot \frac{s'}{x'}) + (K+k) \cdot x = K \cdot s - F_0 \quad (34)$$

5. DYNAMIC MODEL WITH FOUR DEGREES OF FREEDOM WITH INTERNAL SYSTEM DAMPING - VARIABLE -

In the paper (ANTONESCU et al., 1985 a) there is presented a dynamic damping model variable as in the previous paragraph, but with four degrees of mobility.

The hypothesis of the existence of four masses in translational motion is made at the same time (see Figure 3).

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Figure 3a shows the kinematic diagram of the classic distribution mechanism, and in FIG. 3b is shown the corresponding dynamic pattern, with four moving masses, thus with four degrees of freedom.

The way in which the four dynamic masses and the corresponding elastic constants, as well as the corresponding damping, are deduced will be presented in the following paragraph.

The dynamic model with four degrees of freedom (Figure 3) is considered, where the four reduced masses of the driven element (valve) are calculated with the formulas (35).

The mass m_1* is calculated as the mass m_1 (mass of the camshaft) that reduces to the valve axis, that is, this mass m_1, multiplies by the theoretical input speed $\dot{y}_{1c}$, square, and is divided by the square of the valve speed $\dot{x}^2$, the ratio between the cam entry speed $\dot{y}_{1c}$ and valve velocity $\dot{x}$, and rises to square, and this square ratio multiplies by the mass m_1.

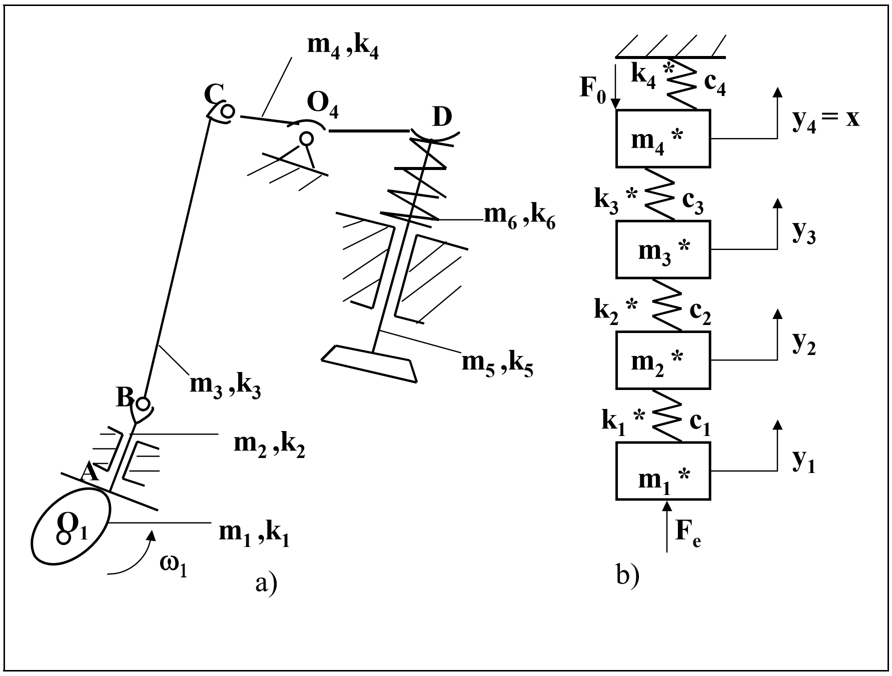

Figure 3: Dynamic model with four degrees of freedom with internal system damping - variable -

As the input speed $\dot{y}_{1c}$ must also be reduced to the axis of the valve, instead of it write down the reduced input velocity to the valve axis $\dot{y}_1$, multiplied by the coulter transmission ratio, i, that is, we have the relation $\dot{y}_{1c} = i.\dot{y}_1$ and the square velocity $\dot{y}_{1c}^2$,

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will be replaced with $i^2 . \dot{y}_1^2$, and will be written down i^2 multiplied to the mass m_1 with m_1'. For mass m_2^*, consider the weight of the tappet, m_2, plus one third of the weight of the pushing rod, m_3, and the corresponding speed $\dot{y}_2$ is practically the dynamic velocity of the tappet reduced to the axis of the valve.

The mass m_3^* corresponds to the pusher rod and consists of two remaining thirds of the pushing rod weight, m_3, plus half of the mass of the stem, m_4; velocity $\dot{y}_3$ is the actual average speed with which the pushing rod moves on the vertical axis reduced to the valve axis, or the speed of the stopper at the point C reduced to the valve axis.

The mass m_4^* is obtained from all the summaries on the side of the valve, ie half the mass of the valve, plus the mass m_5 (which in turn represents the sum of the valve mass and the mass of the valve pan) plus a third of the mass of the valve spring. The speed of the valve (obviously at its axis) was marked with $\dot{x}$.

$$\begin{cases} m_1^* = m_1 \cdot i^2 \cdot (\frac{\dot{y}_1}{\dot{x}})^2 = m_1' \cdot (\frac{\dot{y}_1}{\dot{x}})^2 ; m_2^* = (m_2 + \frac{1}{3} \cdot m_3) \cdot i^2 \cdot (\frac{\dot{y}_2}{\dot{x}})^2 = m_2' \cdot (\frac{\dot{y}_2}{\dot{x}})^2 ; \\ m_3^* = (\frac{2}{3} \cdot m_3 + \frac{1}{2} \cdot m_4) \cdot i^2 \cdot (\frac{\dot{y}_3}{\dot{x}})^2 = m_3' \cdot (\frac{\dot{y}_3}{\dot{x}})^2 ; m_4^* = \frac{1}{2} \cdot m_4 + m_5 + \frac{1}{3} \cdot m_6 = m_4' \end{cases} \quad (35)$$

where i = O_4C / O_4D (see Figure 3) represents the transmission ratio of the culbutor; m_1, m_2, m_3, m_4, m_5, m_6 are in order: the mass of the cam, the stick, the pusher rod, the stem, the valve (with the roller) and the valve spring respectively. The following equivalent elastic constants (see Figure 3) are reduced to the valve (36):

$$K_1^* = \frac{K_1 \cdot K_2}{K_1 + K_2} \cdot i^2 ; K_2^* = K_3 \cdot i^2 ; K_3^* = K_4 ; K_4^* = K_6 \quad (36)$$

where k_1, k_2, k_3, k_4, k_6 are the stiffnesses (elastic constants) of the corresponding elements. The elastic valve constant is not in question. It is noted that F_0 is the external force, known as the spring force of the valve spring, and F_e is the balancing force at the valve, basically the driving force. The influence of moments of mechanical inertia (mass), weight forces and friction forces will be neglected. Following the dynamic equilibrium for each reduced mass in part are written four equations of the form (37-40):

$$K_1^* \cdot (y_1 - y_2) - F_e + m_1^* \cdot \ddot{y}_1 + c_1 \cdot \dot{y}_1 = 0 \quad (37)$$

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$$K_2^* \cdot (y_2 - y_3) - K_1^* \cdot (y_1 - y_2) + m_2^* \cdot \ddot{y}_2 + c_2 \cdot \dot{y}_2 = 0 \tag{38}$$

$$K_3^* \cdot (y_3 - x) - K_2^* \cdot (y_2 - y_3) + m_3^* \cdot \ddot{y}_3 + c_3 \cdot \dot{y}_3 = 0 \tag{39}$$

$$K_4^* \cdot x - K_3^* \cdot (y_3 - x) + F_0 + m_4^* \cdot \ddot{x} + c_4 \cdot \dot{x} = 0 \tag{40}$$

The linear displacements y_1, y_2, y_3, y_4 =x correspond to the reduced masses m_1*, m_2*, m_3*, m_4*.

Assuming that the movement y_1 is known from the motion law $y_1 = y_1$ (□) imposed on the camshaft at the cam design, the displacements y_2, y_3, x and the balance force F_e, ie the motor force F_m, remain unknown.

In this case it is observed that equations (38), (39) and (40) form a system of three equations with three unknowns y_2, y_3, x. After calculating the three displacements from (37), the equilibration force F_e is obtained.

Basically, the system is not linear because, in addition to the unknowns given by the three displacements, we have as extra unknown the speeds and accelerations derived from unknown movements, ie practically unknown will be ten and only four of the system's equations.

$$c = \frac{1}{2} \cdot \frac{dM}{dt} = \frac{\omega_1}{2} \cdot \frac{dM}{d\varphi} \tag{41}$$

For the actual solution of the equation system (37) - (40), the damping coefficients c1, c2, c3, c4 of formula (41), already known from the system with a degree of freedom and the mass system (35), as follows (42-45):

$$c_1 = \frac{1}{2} . \frac{dm_1^*}{dt} = m_1^{'} . (\frac{\dot{y}_1 . \ddot{y}_1}{\dot{\varphi}^2} - \frac{\dot{y}_1^2 . \ddot{\varphi}}{\dot{\varphi}^3}) \tag{42}$$

$$c_2 = \frac{1}{2} . \frac{dm_2^*}{dt} = m_2^{'} . (\frac{\dot{y}_2 . \ddot{y}_2}{\dot{\varphi}^2} - \frac{\dot{y}_2^2 . \ddot{\varphi}}{\dot{\varphi}^3}) \tag{43}$$

$$c_3 = \frac{1}{2} . \frac{dm_3^*}{dt} = m_3^{'} . (\frac{\dot{y}_3 . \ddot{y}_3}{\dot{\varphi}^2} - \frac{\dot{y}_3^2 . \ddot{\varphi}}{\dot{\varphi}^3}) \tag{44}$$

$$c_4 = \frac{1}{2} . \frac{dm_4^*}{dt} = 0 \tag{45}$$

which can also be written in the form (46-49):

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$$c_1 = m_1^{'}.(\frac{\dot{y}_1}{\dot{x}})^2.(\frac{\ddot{y}_1}{\dot{y}_1} - \frac{\ddot{x}}{\dot{x}}) \quad (46)$$

$$c_2 = m_2^{'}.(\frac{\dot{y}_2}{\dot{x}})^2.(\frac{\ddot{y}_2}{\dot{y}_2} - \frac{\ddot{x}}{\dot{x}}) \quad (47)$$

$$c_3 = m_3^{'}.(\frac{\dot{y}_3}{\dot{x}})^2.(\frac{\ddot{y}_3}{\dot{y}_3} - \frac{\ddot{x}}{\dot{x}}) \quad (48)$$

$$c_4 = 0 \quad (49)$$

Using Relationships (46-49) and System (35), Relationships (50-53) can be obtained immediately:

$$c_1.\dot{y}_1 = m_1^{'}.(\frac{\dot{y}_1}{\dot{x}})^2.(\ddot{y}_1 - \frac{\dot{y}_1}{\dot{x}}.\ddot{x}) = m_1^{*}.(\ddot{y}_1 - \frac{\dot{y}_1}{\dot{x}}.\ddot{x}) \quad (50)$$

$$c_2.\dot{y}_2 = m_2^{'}.(\frac{\dot{y}_2}{\dot{x}})^2.(\ddot{y}_2 - \frac{\dot{y}_2}{\dot{x}}.\ddot{x}) = m_2^{*}.(\ddot{y}_2 - \frac{\dot{y}_2}{\dot{x}}.\ddot{x}) \quad (51)$$

$$c_3.\dot{y}_3 = m_3^{'}.(\frac{\dot{y}_3}{\dot{x}})^2.(\ddot{y}_3 - \frac{\dot{y}_3}{\dot{x}}.\ddot{x}) = m_3^{*}.(\ddot{y}_3 - \frac{\dot{y}_3}{\dot{x}}.\ddot{x}) \quad (52)$$

$$c_4.\dot{y}_4 = c_4.\dot{x} = 0 \quad (53)$$

Taking into account relations (50-53), equations (37-40) are rewritten as follows (54-57):

$$K_1^{*}.y_1 - K_1^{*}.y_2 - F_e + 2.m_1^{'}.(\frac{\dot{y}_1}{\dot{x}})^2.\ddot{y}_1 - m_1^{'}.(\frac{\dot{y}_1}{\dot{x}})^3.\ddot{x} = 0 \quad (54)$$

$$-K_1^{*}.y_1 + (K_1^{*} + K_2^{*}).y_2 - K_2^{*}.y_3 + 2.m_2^{'}.(\frac{\dot{y}_2}{\dot{x}})^2.\ddot{y}_2 - m_2^{'}.(\frac{\dot{y}_2}{\dot{x}})^3.\ddot{x} = 0 \quad (55)$$

$$-K_2^{*}.y_2 + (K_2^{*} + K_3^{*}).y_3 - K_3^{*}.x + 2.m_3^{'}.(\frac{\dot{y}_3}{\dot{x}})^2.\ddot{y}_3 - m_3^{'}.(\frac{\dot{y}_3}{\dot{x}})^3.\ddot{x} = 0 \quad (56)$$

$$-K_3^{*}.y_3 + (K_3^{*} + K_4^{*}).x + m_4^{'}.\ddot{x} + F_0 = 0 \quad (57)$$

With the system of equations (54-57), the dynamic model shown in Figure 3 is solved, given that the system is nonlinear and besides the four main unknowns, y_2, y_3, x, F_e, six more unknown $\dot{y}_2, \ddot{y}_2, \dot{y}_3, \ddot{y}_3, \dot{x}, \ddot{x}$ occur, but dependent on each other and also depend on linear displacements, y_2, y_3, and x respectively.

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The system is greatly simplified if we consider the three speeds approximately equal to each other and equal to the known entry speed; In this case, the equation system (54-57) is considerably simplified, taking the form (58-61):

$$K_1^* \cdot y_1 - K_1^* \cdot y_2 - F_e + 2 \cdot m_1^{'} \cdot \dot{y}_1 \dot{x} - m_1^{'} \cdot \ddot{x} = 0 \quad (58)$$

$$-K_1^* \cdot y_1 + (K_1^* + K_2^*) \cdot y_2 - K_2^* \cdot y_3 + 2 \cdot m_2^{'} \cdot \dot{y}_2 \dot{x} - m_2^{'} \cdot \ddot{x} = 0 \quad (59)$$

$$-K_2^* \cdot y_2 + (K_2^* + K_3^*) \cdot y_3 - K_3^* \cdot x + 2 \cdot m_3^{'} \cdot \dot{y}_3 \dot{x} - m_3^{'} \cdot \ddot{x} = 0 \quad (60)$$

$$-K_3^* \cdot y_3 + (K_3^* + K_4^*) \cdot x + m_4^{'} \cdot \ddot{x} + F_0 = 0 \quad (61)$$

6. RESULTS; SOLVING THE DIFFERENTIAL EQUATION

In the paper was presented a dynamic model with a degree of mobility, internal damping of the variable system, which finally leads to the equation (54), which can be writhed in the form (62) and the simplified equation (53), arranged now in form (63).

$$(K+k).x = K.y - k.x_0 - \omega^2.m_S.X^{II} - \omega^2.m_T.y''.\frac{y'}{X^I} \quad (62)$$

$$(K+k).x = K.y - k.x_0 - \omega^2.m_S.X^{II} - \omega^2.m_T.y'' \quad (63)$$

Differential equation (63), ie the simplified form (in which the reduced input velocity imposed by the cam profile y' is equal to the low dynamic velocity, x', both reduced to the valve axis) is used.

6.1. Solving the differential equation, through a particular solution

Equation (63) is written as (64):

$$m_S.\ddot{X} + (K+k).X = K.y - k.x_0 - m_T.\ddot{y} \quad (64)$$

One divides equation (64) with m_S and amplify the straight term with cosωt, thus obtaining the form (65):

$$\ddot{X} + \frac{K+k}{m_S} \cdot X = \frac{K \cdot y - k \cdot x_0 - m_T \cdot \ddot{y}}{m_S \cdot \cos(\omega \cdot t)} \cdot \cos(\omega \cdot t) \quad (65)$$

The following notations (66-67) are used:

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$$p^2 = \frac{K+k}{m_S} \tag{66}$$

$$q = \frac{K.y - k.x_0 - m_T.\ddot{y}}{m_S.\cos(\omega.t)} \tag{67}$$

Equation (65) is written in simplified form (68):

$$\ddot{X} + p^2.X = q.\cos(\omega.t) \tag{68}$$

The particular solution of equation (68) is of the form (69):

$$X = a.\cos(\omega.t) \tag{69}$$

Derivatives 1 and 2 of solution (69) are denoted by (70-71):

$$\dot{X} = -a.\omega.\sin(\omega.t) \tag{70}$$

$$\ddot{X} = -a.\omega^2.\cos(\omega.t) \tag{71}$$

By replacing values (69) and (71) in equation (68), form (72) is obtained:

$$-a.\omega^2.\cos(\omega.t) + p^2.a.\cos(\omega.t) = q.\cos(\omega.t) \tag{72}$$

The characteristic equation is written as (73):

$$a.(p^2 - \omega^2) = q \tag{73}$$

It is explicit a in the form (74):

$$a = \frac{q}{p^2 - \omega^2} \tag{74}$$

Now write the solution X, under the forms (75), (76):

$$X = \frac{q}{p^2 - \omega^2}.\cos(\omega.t) \tag{75}$$

$$X = \frac{K.y - k.x_0 - m_T.\ddot{y}}{m_S.\cos(\omega.t)}.\frac{\cos(\omega.t)}{\frac{K+k}{m_S} - \omega^2} = \frac{K.y - k.x_0 - m_T.\ddot{y}}{K + k - m_S.\omega^2} \tag{76}$$

For a more exact solution, we approximate directly in equation (74), X'' cu y'' cu s'', ie $\ddot{X} = \ddot{y} = \ddot{s}$, and one arrives at the linear equation (77):

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$$X = \frac{K.s - k.x_0 - (m_S + m_T).\ddot{s}}{K + k} = \frac{K.s - k.x_0 - m^*.\ddot{s}}{K + k} \qquad (77)$$

6.2. Solving the differential equation, through a complete private solution

Equation (64) can be written as (78), taking into account coefficients D and D ':

$$m_S.\omega^2.D.x'' + m_S.\omega^2.D'.x' + (K + k).x = K.s - k.x_0 - m_T.\omega^2.(D.s'' + D'.s') \qquad (78)$$

One divides equation (78) with $m_S.\omega^2.D$ and obtain the form (79):

$$x'' + \frac{m_S.\omega^2.D'}{m_S.\omega^2.D}.x' + \frac{K + k}{m_S.\omega^2.D}.x = \frac{K.s - k.x_0 - m_T.\omega^2.(D.s'' + D'.s')}{m_S.\omega^2.D} \qquad (79)$$

The right term is amplified with (cosφ+sinφ) and equation (79) is written as (80):

$$x'' + \frac{D'}{D}.x' + \frac{K + k}{m_S.\omega^2.D}.x = \frac{K.s - k.x_0 - m_T.\omega^2.(D.s'' + D'.s')}{m_S.\omega^2.D.(\cos\varphi + \sin\varphi)}.(\cos\varphi + \sin\varphi) \qquad (80)$$

We note the corresponding coefficients (81-83):

$$a = \frac{D'}{D} \qquad (81)$$

$$b = \frac{K + k}{m_S.D.\omega^2} \qquad (82)$$

$$c = \frac{K.s - k.x_0 - m_T.\omega^2.(D.s'' + D'.s')}{m_S.\omega^2.D.(\cos\varphi + \sin\varphi)} \qquad (83)$$

Equation (80) can now be written as (84):

$$x'' + a.x' + b.x = c.(\cos\varphi + \sin\varphi) \qquad (84)$$

The complete particular solution of equation (84) is of the form (85), and its derivatives according to the angle φ, the derivatives I and II, take the forms (86), respectively (87):

$$x = A.\cos\varphi + B.\sin\varphi \qquad (85)$$

$$x' = -A.\sin\varphi + B.\cos\varphi \qquad (86)$$

$$x'' = -A.\cos\varphi - B.\sin\varphi \qquad (87)$$

Introducing solutions (85-87) in (84) one obtains equation (88):

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$$-A.\cos\varphi - B.\sin\varphi - a.A.\sin\varphi + a.B.\cos\varphi + b.A.\cos\varphi + b.B.\sin\varphi = C.\cos\varphi + C.\sin\varphi \quad (88)$$

We identify the coefficients in the cosine and those in the sin and one obtains a linear system of two equations with two unknown, A and B respectively:

$$\begin{cases}(b-1).A + a.B = c \\ -a.A + (b-1).B = c\end{cases} \quad (89)$$

For the operative solving of the system (89) the first equation increases with a and the second with (b-1), after which B is collected and then determined by A, multiplying the first equation with (b-1) and the second one with -a, after which it collects and obtains the system (90):

$$\begin{cases}A = \dfrac{c}{a^2+(b-1)^2}.(b-1-a) \\ B = \dfrac{c}{a^2+(b-1)^2}.(b-1+a)\end{cases} \quad (90)$$

The solution can now be written as (91), where the coefficients a, b, c are known (81-83):

$$x = \frac{c}{a^2+(b-1)^2}\cdot[(b-1-a)\cdot\cos\varphi + (b-1+a)\cdot\sin\varphi] \quad (91)$$

6.3. Solving the differential equation, with the help of Taylor series developments

Write the relation (92), which expresses the connection between the dynamic displacement of the valve, x, and that imposed by the cam profile, s:

$$x(\varphi) = s(\varphi) + \Delta x(\varphi) \cong s(\varphi + \Delta\varphi) \quad (92)$$

The function s(φ+Δφ) was developed in a Taylor series and retains the first 8 terms of development; now find the relationship (93):

$$\begin{aligned} x = s(\varphi+\Delta\varphi) &= \frac{1}{0!}s(\varphi).(\Delta\varphi)^0 + \frac{1}{1!}s^I(\varphi).\Delta\varphi \\ &+ \frac{1}{2!}.s^{II}(\varphi).(\Delta\varphi)^2 + \frac{1}{3!}.s^{III}(\varphi).(\Delta\varphi)^3 + \frac{1}{4!}.s^{IV}(\varphi).(\Delta\varphi)^4 \\ &+ \frac{1}{5!}.s^{V}(\varphi).(\Delta\varphi)^5 + \frac{1}{6!}.s^{VI}(\varphi).(\Delta\varphi)^6 + \frac{1}{7!}.s^{VII}(\varphi).(\Delta\varphi)^7 \end{aligned} \quad (93)$$

The relationship (93) is also written in the form (94):

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$$x = s + s^{I}.\Delta\varphi + \frac{1}{2}.s^{II}.(\Delta\varphi)^2 + \frac{1}{6}.s^{III}.(\Delta\varphi)^3 + \frac{1}{24}.s^{IV}.(\Delta\varphi)^4 + \frac{1}{120}.s^{V}.(\Delta\varphi)^5 + \frac{1}{720}.s^{VI}.(\Delta\varphi)^6 + \frac{1}{5040}.s^{VII}.(\Delta\varphi)^7 \quad (94)$$

By derivation it obtains x' (relation 95):

$$x^{I} = s^{I} + s^{II}.\Delta\varphi + \frac{1}{2}.s^{III}.(\Delta\varphi)^2 + \frac{1}{6}.s^{IV}.(\Delta\varphi)^3 + \frac{1}{24}.s^{V}.(\Delta\varphi)^4 + \frac{1}{120}.s^{VI}.(\Delta\varphi)^5 + \frac{1}{720}.s^{VII}.(\Delta\varphi)^6 + \frac{1}{5040}.s^{VIII}.(\Delta\varphi)^7 \quad (95)$$

Deriving the second time and get x'', (relation 96):

$$x^{II} = s^{II} + s^{III}.\Delta\varphi + \frac{1}{2}.s^{IV}.(\Delta\varphi)^2 + \frac{1}{6}.s^{V}.(\Delta\varphi)^3 + \frac{1}{24}.s^{VI}.(\Delta\varphi)^4 + \frac{1}{120}.s^{VII}.(\Delta\varphi)^5 + \frac{1}{720}.s^{VIII}.(\Delta\varphi)^6 + \frac{1}{5040}.s^{IX}.(\Delta\varphi)^7 \quad (96)$$

The differential equation used is (62), ie the complete equation, which we write in the form (97), also taking into account the transmission function, D.

$$x = \frac{K \cdot s - k \cdot x_0 - m_S^* \cdot (D \cdot x'' + D' \cdot x') \cdot \omega^2 * 0.001 - m_T^* \cdot (D \cdot s'' + D' \cdot s') \cdot \omega^2 * 0.001 * \frac{s'}{x'}}{K + k} \quad (97)$$

7. DISCUSSION

Dynamic analysis for sinus law, using the relationship (97), based on Taylor series and dynamic-A1 model, with variable internal damping, without considering the mass m1 of the cam.

Using the relation (97) obtained from the differential equation (62) based on the dynamic damping model of the variable system, without considering the mass m1 of the cam, but using Taylor series calculations with the retention of 8 consecutive terms, dynamic (A1).

For this dynamic model (A1) there is a single dynamic diagram (Figure 4).

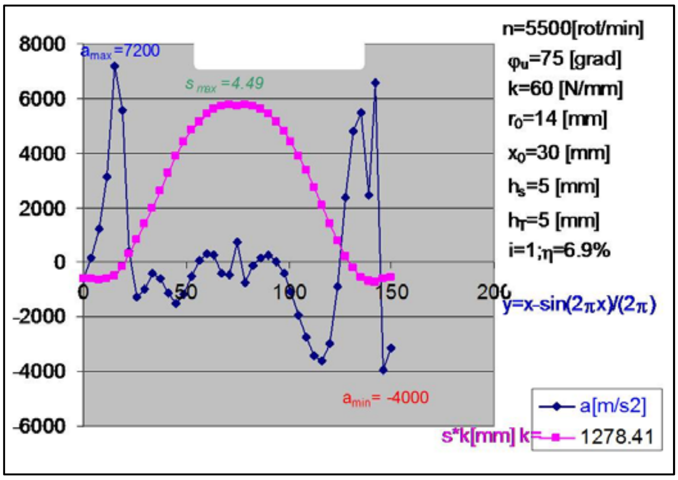

Figure 4: Dynamic analysis using the dynamic A1 model

The SINus law is used, the engine speed, n = 5500 [rpm], equal ascension and descent angles, $\varphi_u=\varphi_c=75^0$, radius of the base circle, r_0 = 14 [mm]. For the maximum stroke of the tappet, h_T, equal to that of the valve, h_S (i = 1), the value of h = 5 [mm] was taken. A spring elastic constant is adopted, k = 60 [N / mm], for a valve spring compression of x_0 = 30 [mm].

Mechanical yield is low (generally in rotary cam and punch mechanisms, mechanical efficiency has low values, and in Module C-classical distribution mechanism these values are even slightly lower), η=6.9%.

The theoretical model presented and used has the advantages of simulating even the fine vibrations of the mechanism.

8. CONCLUSIONS

The development and diversification of road vehicles and vehicles, especially of cars, together with thermal engines, especially internal combustion engines (being more compact, robust, more independent, more reliable, stronger, more dynamic etc.)., has also forced the development of devices, mechanisms, and component assemblies at an alert pace. The most studied are power and transmission trains.

The four-stroke internal combustion engine (four-stroke, Otto or Diesel) comprises in most cases (with the exception of rotary motors) and one or more camshafts, valves, valves, and so on.

The classical distribution mechanisms are robust, reliable, dynamic, fast-response, and although they functioned with very low mechanical efficiency, taking much of the engine power and effectively causing additional pollution and increased

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fuel consumption, they could not be abandoned until the present. Another problem was the low speed from which these mechanisms begin to produce vibrations and very high noises.

Regarding the situation realistically, the mechanisms of cam casting and sticking are those that could have produced more industrial, economic, social revolutions in the development of mankind. They have contributed substantially to the development of internal combustion engines and their spreading to the detriment of external combustion (Steam or Stirling) combustion engines.

The problem of very low yields, high emissions and very high power and fuel consumption has been greatly improved and regulated over the past 20-30 years by developing and introducing modern distribution mechanisms that, besides higher yields immediately deliver a high fuel economy) also performs optimal noise-free, vibration-free, no-smoky operation, as the maximum possible engine speed has increased from 6000 to 30000 [rpm].

The paper tries to provide additional support to the development of distribution mechanisms so that their performance and the engines they will be able to further enhance.

Particular performance is the further increase in the mechanical efficiency of distribution systems, up to unprecedented quotas so far, which will bring a major fuel economy.

The paper presents a dynamic model that works with variable internal damping, applicable directly to rigid memory mechanisms. If the problem of elasticity is generally solved, the problem of system damping is not clear and well-established. It is usually considered a constant "c" value for the internal damping of the system and sometimes the same value c and for the damping of the elastic spring supporting the valve.

However, the approximation is much forced, as the elastic spring damping is variable, and for the conventional cylindrical spring with constant elasticity parameter (k) with linear displacement with force, the damping is small and can be considered zero. It should be specified that damping does not necessarily mean stopping (or opposition) movement, but damping means energy consumption to brake the motion (rubber elastic elements have considerable damping, as are hydraulic dampers).

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Metal helical springs generally have a low (negligible) damping. The braking effect of these springs increases with the elastic constant (the k-stiffness of the spring) and the force of the spring (P_0 or F_0) of the spring (in other words with the arc static arrow, $x_0=P_0/k$). Energy is constantly changing but does not dissipate (for this reason, the yield of these springs is generally higher).

The paper presents a dynamic model with a degree of freedom, considering internal damping of the system (c), damping for which it is considered a special function. More precisely, the cushioning coefficient of the system (c) is defined as a variable parameter depending on the reduced mass of the mechanism (m* or J reduced) and the time, ie, c depends on the derivative of m reduced in time.

The equation of the differential movement of the mechanism is written as the movement of the valve as a dynamic response. Dynamic analysis for sinus law, using the relationship (97), based on Taylor series and dynamic-A1 model, with variable internal damping, without considering the mass m1 of the cam.

Using the relation (97) obtained from the differential equation (62) based on the dynamic damping model of the variable system, without considering the mass m_1 of the cam, but using Taylor series calculations with the retention of 8 consecutive terms, dynamic (A1). For this dynamic model (A1) there is a single dynamic diagram (Figure 4).

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11. NOMENCLATURE

J^*	is the moment of inertia (mass or mechanical) reduced to the camshaft
J^*_{Max}	is the maximum moment of inertia (mass or mechanical) reduced to the camshaft
J^*_{min}	is the minimum moment of inertia (mass or mechanical) reduced to the camshaft
J^*_m	is the average moment of inertia (mass or mechanical, reduced to the camshaft)
$J^{*\prime}$	is the first derivative of the moment of inertia (mass or mechanical, reduced to the camshaft) in relation with the φ angle
η_i	is the momentary efficiency of the cam-pusher mechanism
η	is the mechanical yield of the cam-follower mechanism
τ	is the transmission angle
δ	is the pressure angle
s	is the movement of the pusher
h	is the follower stroke $h=s_{max}$
s'	is the first derivative in function of φ of the tappet movement, s
s''	is the second derivative in raport of φ angle of the tappet movement, s
s'''	is the third derivative of the tappet movement s, in raport of the φ angle
x	is the real, dynamic, movement of the pusher
x'	is the real, dynamic, reduced tappet speed
x''	is the real, dynamic, reduced tappet acceleration
&&	is the real, dynamic, acceleration of the tappet (valve).
$v_\tau \equiv$ &	is the normal (cinematic) velocity of the tappet
$a_\tau \equiv$ &&	is the normal (cinematic) acceleration of the tappet
φ	is the rotation angle of the cam (the position angle)
K	is the elastic constant of the system
k	is the elastic constant of the valve spring
x_0	is the valve spring preload (pretension)
m_c	is the mass of the cam
m_T	is the mass of the tappet
ω_m	the nominal angular rotation speed of the cam (camshaft)
n_c	is the camshaft speed
$n=n_m$	is the motor shaft speed $n_m=2n_c$
ω	is the dynamic angular rotation speed of the cam
ε	is the dynamic angular rotation acceleration of the cam
r_0	is the radius of the base circle
ρ=r	is the radius of the cam (the position vector radius)
θ	is the position vector angle
$x=x_c$ and $y=y_c$	are the Cartesian coordinates of the cam

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D	is the dynamic coefficient
$\dot{D}$	is the derivative of D in function of the time
D'	is the derivative of D in function of the position angle of the camshaft, φ
Fm	is the motor force
Fr	is the resistant force.

12. AUTHORS' CONTRIBUTION

All the authors have contributed equally to carry out this work.

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